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# CO<sub>2</sub> and R410A flow boiling heat transfer and pressure drop at low temperatures in a horizontal smooth tube

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## ABSTRACT

Flow boiling heat transfer coefficient and pressure drop are investigated in the horizontal smooth tube of 6.1 mm inner diameter for CO<sub>2</sub>, R410A, and R22. Flow boiling heat transfer coefficients are measured at the constant wall temperature conditions, while pressure drop measurement is carried out at adiabatic conditions. This research is performed at evaporation temperatures of -15 and -30 °C, mass flux from 100 to 400 kg/m<sup>2</sup> s, and heat flux from 5 to 15 kW/m<sup>2</sup> for vapor qualities ranging from 0.1 to 0.8. The measured R410A heat transfer coefficients are compared to other published data. The comparison of heat transfer coefficients for CO<sub>2</sub>, R410A, and R22 is presented at various heat fluxes, mass fluxes, and evaporation temperatures. The Wattelet et al. (1994), and Gungor and Winterton (1986) correlations give the best agreement with the measured heat transfer coefficients for CO<sub>2</sub> and R410A. Pressure drop for CO<sub>2</sub>, R410A, and R22 at various mass fluxes, evaporation temperatures and qualities is presented in this paper. The Müller-Steinhagen and Heck (1986), and Friedel (1979) correlation can predict the measured pressure drop relatively well.

## 1. INTRODUCTION

CO<sub>2</sub> has been seriously considered as an alternate refrigerant for HFCs. R410A is one of the most widely used HFCs as a replacement of CFCs and HCFCs at air-conditioning and refrigeration systems, while R22 is probably the best described fluid of all three. As the application of CO<sub>2</sub> to real systems increases, the accurate measurements and the comparison of the flow boiling heat transfer coefficients and pressure drop for CO<sub>2</sub> and R410A are required over wide ranges of operating conditions such as low temperature applications.

The flow boiling heat transfer coefficients for CO<sub>2</sub> at low evaporation temperature were presented by Bredeesen et al. (1997), Høgaard Knudsen and Jensen (1997), and Park and Hrnjak (2005). Bredeesen et al. (1997) presented CO<sub>2</sub> boiling heat transfer coefficients and pressure drop in a 7.0 mm inner diameter tube for various mass fluxes, heat fluxes and evaporation temperatures of -25, -10, and 5 °C. Their data showed that dryout occurred only at 5 °C. Based on their measured results, they concluded that nucleate boiling is a more important mechanism of heat transfer for CO<sub>2</sub> than other refrigerants. Høgaard Knudsen and Jensen (1997) measured heat transfer coefficients in a 10.06 mm tube at -25 and -10 °C. Park and Hrnjak (2005) showed the heat transfer coefficients in a 6.1 mm inner diameter tube at -30 and -15 °C for various mass fluxes and heat fluxes. Their data also confirmed the former interpretation for the major contribution of nucleate boiling in flow boiling heat transfer. Even though data for the heat transfer coefficients and pressure drop for R410A are extensive, a small amount of data was presented for low temperature evaporation conditions in open literature. Kim et al. (2002) measured the R410A flow boiling heat transfer coefficients and pressure drop in 7 and 9.52 mm smooth and micro tubes at -15, -5 and 5 °C. Greco and Vanoli (2005a) presented heat transfer coefficients in 6 mm tube at the evaporation temperatures range from -14.9 to 14.3 °C. From the literature review, it can be found that the flow boiling heat transfer and pressure drop data for CO<sub>2</sub> and R410A at low temperature are very limited. Although Greco and Vanoli (2005a) stated that the R410A heat transfer coefficients were always higher at low evaporation temperatures than R404A, a common refrigerant in refrigeration systems, the R410A heat transfer coefficients at lower evaporation temperatures than -15 °C were not presented. Due to the scarcity of data for CO<sub>2</sub> and R410A heat transfer coefficient and pressure drop data at low temperature conditions, it was difficult to compare the two refrigerants which can provide the useful criteria for

determining the CO<sub>2</sub> as a prospective replacement of conventional refrigerants at low temperature applications. In this study, CO<sub>2</sub> and R410A heat transfer coefficients and pressure drop were measured at various test conditions at the low evaporation temperatures of  $-15$  and  $-30$  °C. Also, R22 heat transfer coefficients and pressure drop were measured and compared with CO<sub>2</sub> and R410A at the evaporation temperature of  $-15$  °C. All measurements were performed in the same test facilities, which excluded the effect of facility on heat transfer coefficients and pressures drops of the refrigerants.

## 2. EXPERIMENTAL FACILITY AND TEST CONDITIONS

Figure 1 shows a schematic of the experiment facility. The test facility has 2 independent loops; one is for a refrigerant, CO<sub>2</sub> or R410A, and the other is for the secondary fluid, HFE7100. The refrigerant loop consists of a gear pump, mass flow meter, calorimeter, test section, visualization section, control heater, receiver and subcooler. Liquid refrigerant is pumped by the gear pump to the calorimeter. The calorimeter heats the subcooled liquid refrigerant to a desired quality at the inlet of the test section and visualization section is located after the test section. Horizontal and vertical pressure drop is measured by differential pressure transducers at adiabatic conditions. The control heater adds heat to maintain a desired saturation temperature in the test section. Then the refrigerant condenses in a plate heat exchanger connected to the R404A cooling unit. To avoid cavitation in the pump, the refrigerant passes through the receiver and enters the subcooler. The other loop is for the single phase secondary fluid HFE7100. HFE subcools the refrigerant in the subcooler and then adds heat in the test section. In order to perform both operations, HFE is chilled in a heat exchanger linked with the R404A cooling unit and is also heated by the HFE heater to provide evaporating conditions in the test section.

The test section consists of the test tube, brass jacket, and tube circuit for HFE as presented at Figure 1. The inside and outside diameters of the test tube are 6.1 and 9.6 mm, respectively. The test tube is made of copper with a heated length of 150 mm and the heated region is surrounded by a brass jacket which consists of two half-cylinder pieces. The secondary fluid, HFE, flows around this jacket in the tube circuit. In order to provide the uniform temperature condition, two half-cylinder shaped brass pieces are located between the HFE circuit and the test tube. The brass jacket unifies normally low temperature glide of the secondary fluid, HFE, used here for heating. All gaps among the two brass pieces and the test tube are filled with high thermal conductivity paste to reduce the contact thermal resistance while the upper and lower parts of brass jackets are tightened with two metal band clamps. Thermocouples are placed at the top, bottom, and sides along 3 locations of test section. The thermocouples were attached in grooves carved on the tube surface with thin solder. The remaining portion of the grooves is filled with high thermal conductivity paste. The thermocouples are equally spaced along the axis of the test section at an interval of 50 mm starting 25 mm from the inlet of the heated section. As a result, the temperatures at 12 points on the tube surface are measured and their average values are used to calculate the heat transfer coefficient.

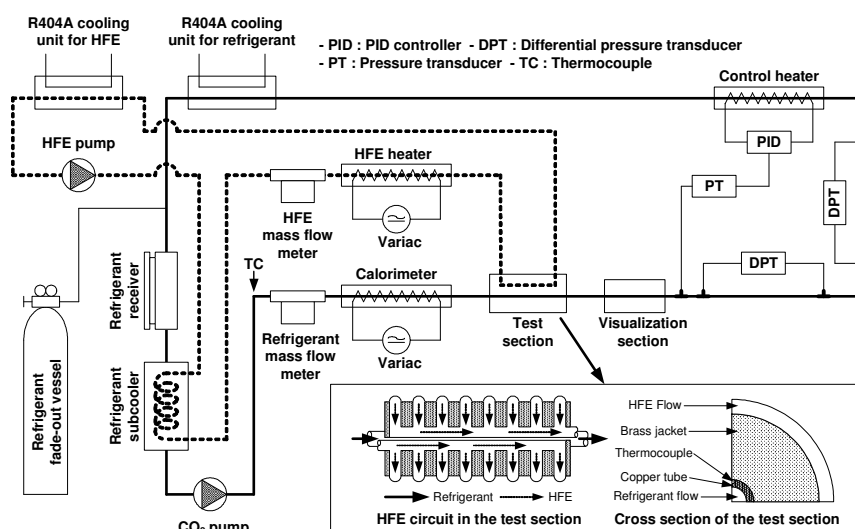


Figure 1. Simplified schematics of test facility and test section

T-type thermocouples with a calibrated accuracy of  $\pm 0.10$  °C are used to measure the refrigerant temperature and wall temperature on the test tube. The absolute pressure of CO<sub>2</sub> is determined by a pressure transducer with an uncertainty of  $\pm 3.4$  kPa and pressure drop is evaluated by differential pressure transducers with the accuracy of  $\pm 0.086$  kPa. The refrigerant mass flow rate is measured by a mass flow meter with an accuracy of  $\pm 0.1\%$  of the reading. Electrical power inputs to the calorimeter and HFE heater are measured with watt transducers which have 0.2% reading accuracy.

Flow boiling heat transfer coefficients for CO<sub>2</sub> and R410A are measured with a variation of saturation temperature, mass flux, heat flux and quality. The evaporation temperature is set at  $-15$  and  $-30$  °C, while the mass flux is varied at 100, 200, and 400 kg/m<sup>2</sup> s with the heat flux variation of 5, 10, and 15 kW/m<sup>2</sup>. The refrigerant quality at the test section inlet is controlled from 0.1 to 0.8.

### 3. DATA REDUCTION

In order to obtain an average heat transfer coefficient, the heat transfer rate to the refrigerant which is CO<sub>2</sub> or R410A in this study,  $\dot{Q}_{ref}$ , is determined as shown in the following equation.

$$\dot{Q}_{ref} = (\dot{m}C_p)_{HFE} \cdot (T_{HFE,i} - T_{HFE,o}) + \dot{Q}_{Amb} - \dot{Q}_{Cond} \quad (1)$$

$$h = \frac{\dot{Q}_{ref}}{A_{surf,w} \cdot (T_{Wall} - T_{CO_2})} \quad (2)$$

As presented at the first term of right hand side of Eq. (1), the heat transfer rate from secondary fluid was determined from the HFE specific heat, mass flow rate, and temperature difference between the inlet and outlet of the test section. The heat exchange rate with the environment,  $\dot{Q}_{Amb}$ , was obtained in a calibration experiment when an electrical heater was inserted in the test section while the power was carefully measured and presented as a function of the overall heat transfer coefficient of the test section and the log mean temperature difference between the HFE and the ambient air. The axially transferred conduction heat loss through the pipe,  $\dot{Q}_{Cond}$ , is estimated by a finite element code. As presented in Eq. (2), the average heat transfer coefficient,  $h$ , is determined from the calculated heat transfer rate to refrigerant,  $\dot{Q}_{ref}$ , measured average tube wall temperature, the test tube geometry, and the refrigerant saturation temperature calculated from the measured saturation pressure. Data regression and determination of refrigerant properties are performed using Engineering Equation Solver (2005).

The uncertainty of the heat transfer coefficient occurs due to the uncertainties of the independent measured parameters; temperature, pressure, mass flow, and electrical power input as presented earlier. The uncertainty propagation of heat transfer coefficient is evaluated based on Moffat (1988). The uncertainty is within the range of 8–20% of the measured heat transfer coefficients. Each measurement uncertainty is shown as a vertical error bar in the figures.

## 4. RESULTS AND DISCUSSION

### 4.1 Comparison with other data at similar test conditions

Figure 2 shows the comparison of heat transfer coefficients for R410A in this study with Kim et al. (2002), and Greco and Vanoli (2005a) at similar test conditions. The flow boiling heat transfer coefficients measured by Kim et al. (2002) are reasonably consistent with the obtained heat transfer coefficients in this study because their heat transfer coefficients for mass flux, 164 kg/m<sup>2</sup> s are between the heat transfer coefficients in this study for the mass flux of 100 and 200 kg/m<sup>2</sup> s at an identical evaporation temperature and heat flux condition. The heat transfer coefficients presented by Greco and Vanoli (2005a) are lower than the coefficients in this study at similar measurement conditions. Greco and Vanoli (2005b) commented that their measured heat transfer coefficients at the test condition presented in Figure 2 were obviously lower than heat transfer coefficients predicted by most of

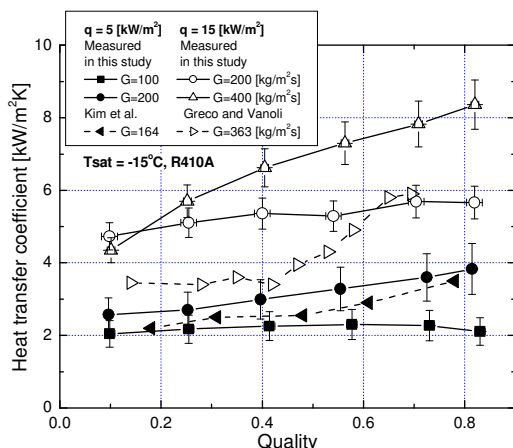


Figure 2. Comparison of heat transfer coefficients for R410A in this study to Kim et al. (2002), and Greco and Vanoli (2005)

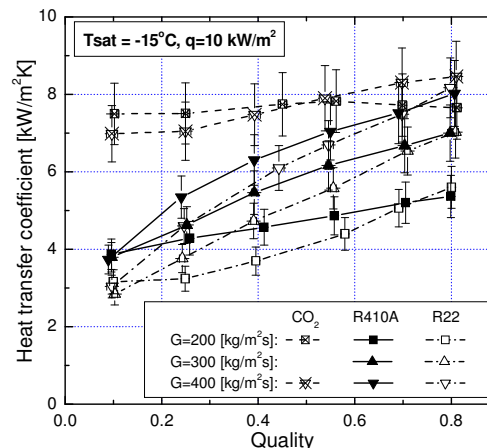


Figure 3. Heat transfer coefficient comparison for CO<sub>2</sub>, R410A and R22 with respect to mass fluxes and quality at the evaporation temperature of  $-15^{\circ}\text{C}$  and the heat flux of  $10\text{ kW/m}^2$

general correlations in open literatures. The comparison of the measured CO<sub>2</sub> flow boiling heat transfer coefficients in this test facility with other data proposed by Bredesen et al. (1997), and Høgaard Knudsen and Jensen (1997) was presented by Park and Hrnjak (2005).

#### 4.2 Flow boiling heat transfer coefficients

Figure 3 shows the heat transfer coefficient comparison for CO<sub>2</sub>, R410A and R22 at an evaporation temperature of  $-15^{\circ}\text{C}$ , a heat flux of  $10\text{ kW/m}^2$ , and with a mass flux variation from 200 to  $400\text{ kg/m}^2\text{ s}$ . The presented data was obtained at the same test facility and test conditions with charging each refrigerant, CO<sub>2</sub>, R410A, and R22. At low quality regions below 0.3, the heat transfer coefficients of CO<sub>2</sub> are much higher than those of R410A and R22 for all mass fluxes, and as the quality and mass flux increase the improvement of the heat transfer coefficient decreases. The heat transfer coefficients of R410A are higher than those of R22, even though the improvement is not significant. As presented in Figure 3, the heat transfer coefficient for CO<sub>2</sub> is nearly independent of vapor quality, whereas the heat transfer coefficient for R410A and R22 significantly increases as the quality and mass flux increase. This trend can be explained by the difference of the density ratio of liquid to vapor for CO<sub>2</sub>, R410A and R22. Convective boiling is usually enhanced by the increasing of the average velocities of liquid and vapor as the quality increases. As the density ratio of liquid to vapor decreases, there is a smaller variation in the convective boiling heat transfer coefficient as quality increases due to the smaller change in the liquid and vapor average velocities. The density ratio of liquid to vapor for CO<sub>2</sub> is 16.60, and that for R410A and R22 are 66.70 and 103.1 at  $-15^{\circ}\text{C}$ , respectively. Consequently, CO<sub>2</sub> flow boiling heat transfer coefficients are almost independent of quality due to the combination of relatively smaller change of convective heat transfer with respect to quality and high nucleate boiling values. However, CO<sub>2</sub> flow boiling heat transfer coefficients have a positive slope for a mass flux of  $400\text{ kg/m}^2\text{ s}$  because of the increase of convective boiling contribution due to the increase in the mass flux.

Figures 4 and 5 show the flow boiling heat transfer coefficients for CO<sub>2</sub> and R410A with the change of mass flux and heat flux at the evaporation temperatures of  $-15$  and  $-30^{\circ}\text{C}$ , respectively. Heat transfer coefficients for CO<sub>2</sub> are always higher than for R410A at every identical test condition mainly due to the higher nucleate boiling contribution. Unlike the heat transfer coefficients of conventional refrigerant, CO<sub>2</sub> heat transfer coefficients for a higher mass flux,  $400\text{ kg/m}^2\text{ s}$ , are lower than those for a lower mass flux,  $200\text{ kg/m}^2\text{ s}$ , at low quality regions. The unexpected decrease in heat transfer with increasing mass flux was also reported by Bredesen et al. (1997) at low qualities, less than 0.3. Thome and Hajal (2004) cited that the increase of mass flux does not always give a higher heat transfer coefficient for CO<sub>2</sub> based on their prediction results. However, the decrease of heat transfer coefficients with increasing mass flux was not measured for R410A. Heat transfer coefficients for R410A show the higher heat transfer coefficients for higher mass flux and quality at every test conditions in Figures 4 and 5. This trend demonstrates that the convective boiling is significantly active heat transfer mechanism for R410A. Whereas, the enhancement of heat transfer coefficients with the increase of mass flux and quality is not significant for CO<sub>2</sub> because of the nucleate boiling dominance on CO<sub>2</sub> flow boiling heat transfer.

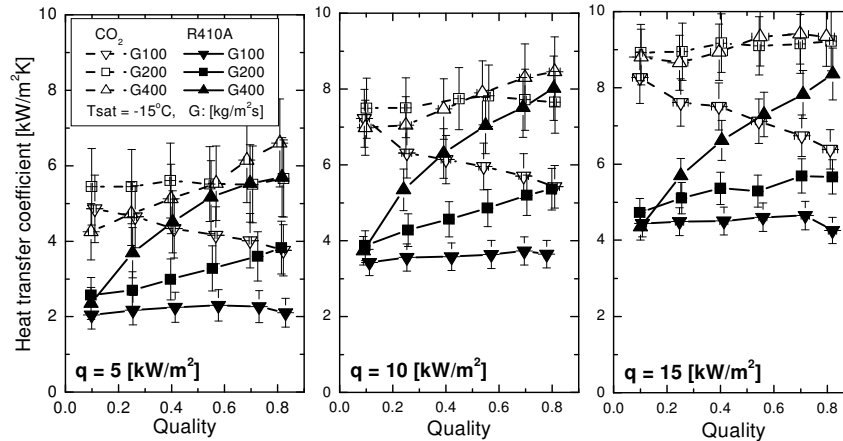


Figure 4. Flow boiling heat transfer coefficients for CO<sub>2</sub> and R410A with the change of mass flux and heat flux at an evaporation temperature of  $-15^{\circ}\text{C}$

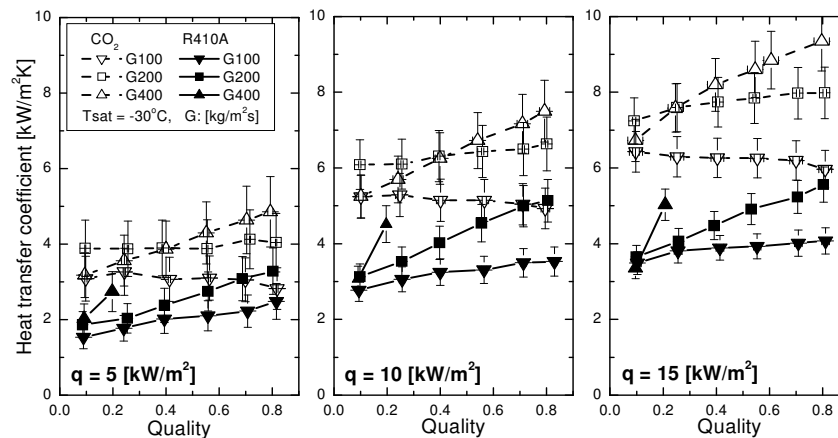


Figure 5. Flow boiling heat transfer coefficients for CO<sub>2</sub> and R410A with the change of mass flux and heat flux at an evaporation temperature of  $-30^{\circ}\text{C}$

For mass flux,  $100\text{ kg/m}^2\text{ s}$ , heat transfer coefficients for CO<sub>2</sub> and R410A show different trends with the increase of quality. Especially for the evaporation temperature of  $-15^{\circ}\text{C}$ , heat transfer coefficients for CO<sub>2</sub> show decreasing trend and those for R410A present the steady values with the increase of quality for the mass flux of  $100\text{ kg/m}^2\text{ s}$ . This trend can be explained by the difference of flow patterns which were observed in this study. According to the visualization, the CO<sub>2</sub> flow pattern for a mass flux of  $100\text{ kg/m}^2\text{ s}$  shows a stratified flow pattern at vapor qualities above 0.4 and 0.3 for evaporation temperatures of  $-15$  and  $-30^{\circ}\text{C}$ , respectively. At the lower vapor qualities where the stratified flow occurs, a flow pattern is a slug and stratified flow. The stratified flow means that there is insufficient liquid film on the top part of a tube. The deficiency of a liquid film gets more severe as the quality increases. Under stratified flow conditions, nucleate boiling cannot be initiated on the top part of the tube wall, which results in the lower flow boiling heat transfer coefficients. The R410A flow patterns for a mass flux of  $100\text{ kg/m}^2\text{ s}$  demonstrated a “slug + stratified” flow for low qualities and an annular flow for mid and high quality regions. As a result, nucleate boiling can occur actively on the tube surface and flow boiling coefficients maintained over the entire quality range up to 0.8.

The flow boiling heat transfer coefficients for CO<sub>2</sub> and R410A at the evaporation temperature of  $-15^{\circ}\text{C}$  are always higher than those at  $-30^{\circ}\text{C}$  as presented in Figures 4 and 5. This trend is mainly because the nucleate boiling heat transfer is reduced with decreasing evaporation temperature. The reduction of the nucleate boiling heat transfer is related to a decrease of the reduced pressure, which is an important parameter to determine the intensity of nucleate boiling in the Gorenflo (1993) correlation. The reduced pressures for CO<sub>2</sub> at  $-15$  and  $-30^{\circ}\text{C}$  are 0.310 and 0.194, respectively. Also, those for R410A at  $-15$  and  $-30^{\circ}\text{C}$  are 0.0976 and 0.0549, respectively. Evaporation

temperatures influence more on CO<sub>2</sub> heat transfer coefficients than on R410A because nucleate boiling heat transfer is more dominant for CO<sub>2</sub>.

Based on the comparison between the measured and calculated heat transfer coefficients, Park and Hrnjak (2005) presented that the Gungor and Winterton (1986) correlation could predict the CO<sub>2</sub> flow boiling heat transfer coefficients relatively well at low evaporation temperatures. They commented that the comparison with the general correlations, which were developed without the heat transfer coefficient database of CO<sub>2</sub>, was still valuable for CO<sub>2</sub> especially at low evaporation temperatures because its thermophysical properties at -15 °C are similar to those of conventional refrigerants such as R22 at 10 °C. In their study, the Gungor and Winterton (1996) correlation tended to overpredict for a mass flux of 400 kg/m<sup>2</sup> s and underpredict for 100 and 200 kg/m<sup>2</sup> s. For R410A, the Wattelet et al. (1994) correlation and the Gungor and Winterton (1996) can predict the measured heat transfer coefficients relatively well.

### 4.3 Two-phase flow pressure drop

The two-phase flow pressure drop for inside tubes can be considered as the sum of three contributions: the static, momentum, and friction pressure drop. In this study, the pressure drop for CO<sub>2</sub>, R410A and R22 was measured at horizontal and adiabatic conditions. As a result, friction contribution is the main pressure drop mechanism in this study. The accuracy of pressure drop measurement is ±0.086kPa and the measured pressure drop is presented in Figure 6. For all refrigerants, pressure drop increases with the increase of mass flux. Also, they rise with the reduction of evaporation temperature due to the increase of liquid viscosity and the decrease of vapor density. Figure 6 shows that the R410A pressure drop is smaller than R22, and CO<sub>2</sub> pressure drop is much lower than the R22 and R410A at an identical condition. Table 1 presents the bias error and absolute average deviation of the predicted values with some general correlations from the measured pressure drop in this study. For CO<sub>2</sub>, The Müller-Steinhagen and Heck (1986) correlation can calculate the measured pressure drop relatively well with a bias error and absolute average deviation of -5.29% and 19.2%, respectively. Also, the Friedel (1979) correlation can give good agreement with the measured pressure drop. For R410A, the pressure drop can be estimated relatively well by Müller-Steinhagen and Heck (1986) with a bias error and absolute average deviation of -11.3% and 22.6%, respectively. The Friedel (1979) correlation can calculate the measured pressure drop with a bias error and absolute average deviation of 5.07% and 31.1%, respectively. From Table 1, it can be concluded that the pressure drop for CO<sub>2</sub> and R410A can be estimated relatively well by the Müller-Steinhagen and Heck (1986), and Friedel (1979) correlations. Ould Didi et al. (2002) presented that these two correlations were good models to predict the pressure drop for annular flow patterns. The annular flow pattern was observed at more than 70% of the pressure drop measurement conditions in this study.

Table 2. Comparison of experimental data for CO<sub>2</sub> and R410A flow boiling heat transfer coefficients in this study with some general correlations

Correlations	CO <sub>2</sub>		R410A	
	Bias <sup>a</sup> error	AAD <sup>b</sup>	Bias error	AAD
Friedel (1979)	22.8	32.5	5.07	31.1
Müller-Steinhagen and Heck (1986)	-5.29	19.2	-11.3	22.6
Lockhart and Martinelli (1949)	113	115	29.2	50.4
Chisholm (1973)	76.3	78.3	86.2	86.7
Grönnerud (1979)	53.9	56.4	20.4	27.5

<sup>a</sup>Bias error, %:  $\frac{1}{N} \sum [(h_{predicted} - h_{measured}) / h_{measured}] \times 100$

<sup>b</sup>Absolute average deviation (AAD) %:

$$\frac{1}{N} \sum |[(h_{predicted} - h_{measured}) / h_{measured}]| \times 100$$

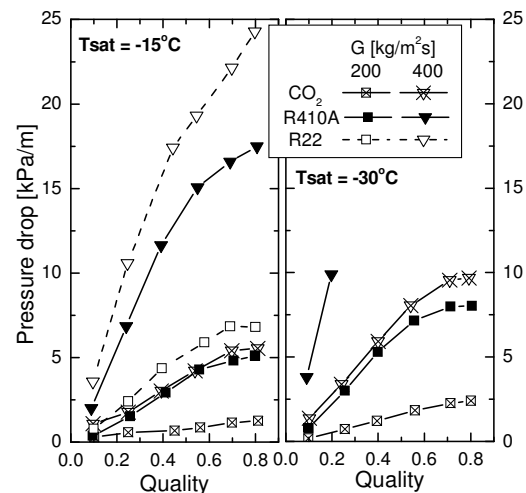


Figure 6. Pressure drop of adiabatic two-phase flow for CO<sub>2</sub>, R410A and R22 with the change of mass flux, quality at the evaporation temperatures of -15 and -30 °C

## 5. SUMMARY AND CONCLUSIONS

The investigation of flow boiling heat transfer coefficients and pressure drop is performed in the horizontal smooth tube of 6.1 mm inner diameter for CO<sub>2</sub>, R410A, and R22 at evaporation temperatures of –15 and –30 °C, mass flux from 100 to 400 kg/m<sup>2</sup> s, and heat flux from 5 to 15 kW/m<sup>2</sup> for vapor qualities ranging from 0.1 to 0.8. Flow boiling heat transfer for CO<sub>2</sub> is much higher than those for R410A and R22 especially at low quality ranges for an identical heat flux, mass flux and evaporation temperature. The lower molecular weight and the higher reduced pressure of CO<sub>2</sub> than those of the other refrigerants result in higher flow boiling heat transfer coefficients by enhancing the nucleate boiling heat transfer contribution. CO<sub>2</sub> heat transfer coefficients show the trend of nucleate boiling dominance in heat transfer mechanism as the strong dependence of heat fluxes and the weak influence from mass flux and quality change. Whereas, nucleate and convective boiling heat transfer mechanisms are active for R410A flow boiling heat transfer because R410A heat transfer coefficients are affected by the change of heat flux, mass flux and quality. The Wattelet et al. (1994), and Gungor and Winterton (1986) correlations can estimate the measured heat transfer coefficients for CO<sub>2</sub> and R410A relatively well. Pressure drop for CO<sub>2</sub> is much lower than those for R410A and R22 mainly due to the higher vapor density of CO<sub>2</sub>. The Müller-Steinhagen and Heck (1986), and Friedel (1979) correlation can predict the measured pressure drop relatively well.

This study indicates that CO<sub>2</sub> has better heat transfer and pressure drop characteristics than the conventional refrigerants of R22 and R410A. From the heat transfer aspect, the advantage of CO<sub>2</sub> is noticeable especially at low temperature applications because dryout does not occur. As a result, the high heat transfer coefficients can be used over the wide range of quality. Also, CO<sub>2</sub> pressure drop is much lower than conventional refrigerants, which means that the applications of CO<sub>2</sub> in refrigeration systems can reduce power input to circulate the working fluid.

## NOMENCLATURE

A	area	(m <sup>2</sup> )	<b>Subscripts</b>	
C <sub>p</sub>	specific heat	(J/kg K)	Amb	ambience
G	mass flux	(kg/m <sup>2</sup> s)	Cond	conduction
h	heat transfer coefficient	(W/m <sup>2</sup> K)	HFE	secondary fluid
$\dot{m}$	mass flow rate	(kg/s)	i	inlet
q	heat flux	(kW/m <sup>2</sup> )	o	outlet
$\dot{Q}$	heat transfer rate	(W)	ref	refrigerant
T	temperature	(°C or K)	sat	saturation

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